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EFFECT OF INTERNAL HEAT EXCHANGER ON PERFORMANCE OF TRANSCRITICAL CO_2 SYSTEMS WITH EJECTOR

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ABSTRACT

The objective of this paper is to explore whether the use of an internal heat exchanger can further improve the performance of transcritical ejector systems with CO_2 as the working fluid. Instead of using a simplified thermodynamic cycle analysis, the approach taken here is based on a more elaborated and experimentally validated system model for a real mobile air-conditioning system for a typical mid-sized car. However, the modeling of the ejector within the system model was still based on some idealized assumptions. The results indicate that the use of an ejector significantly increases the performance compared to systems without ejector and without IHX. The performance improvement effect of the ejector is reduced when the conventional system has an IHX. However, in comparison to a conventional system with IHX, the utilization of an IHX in the ejector system yields less performance increase than the ejector system not having an IHX. The high-side pressure correlations derived and presented here for all four systems can be used in further experimental and modeling work to maximize the cooling capacity and COP, respectively.

1. INTRODUCTION

Besides non-isentropic compression and heat transfer associated with temperature glide and pressure drop, the adiabatic Joule-Thompson throttling process taking place in a conventional expansion device represents a major inefficiency of every air-conditioning system equipped with such a device. HVAC engineers have been constantly trying to devise ways and methods to effectively reduce the throttling related exergy destruction. Some designs include systems comprising multistage expansion with flash gas removal and expander machines. Another device, yet not as well known, theoretically capable of overcoming expansion losses is the refrigerant ejector, invented almost a century ago (Gay, 1931). The ejector principle is based on the isentropic conversion process of pressure related flow work contained in the driving (motive) fluid stream into kinetic energy. The velocity increase associated with low pressure in the throat of the device is used to entrain refrigerant exiting the evaporator by momentum exchange. The following diffuser section of the ejector then isentropically re-compresses the refrigerant by slowing down the mixed high-speed fluid stream. The resulting compression ratio in the compressor is reduced in comparison to a conventional system and thus the ejector refrigeration system theoretically yields dramatically higher system COPs. However, the open literature available for ejector experiments carried out during the last decades with CFCs / HCFCs / HFCs as working fluids reports COP increases in the range of approximately only 5% (e.g. Harrell and Kornhauser (1995) for R134a), mainly arising from the difficulties in designing ejectors operating efficiently at different ambient conditions.

The recent re-discovery of carbon dioxide as a natural refrigerant, with promising thermodynamic properties, led to a renaissance of the ejector refrigeration principle (Denso Corporation, 2002). Up to this point, the high refrigerant pressures at the compressor discharge required in transcritical CO_2 systems were always considered to be somewhat of a drawback. However, the ejector principle applied to a transcritical CO_2 system can make perfect use of the high flow work energies stored in the supercritical motive flow, such that there is only little doubt that the resulting COP improvements will substantially exceed those of CFC / HCFC / HFC ejector systems.

The objective of this paper is to assess the performance improvement potentials of different CO_2 ejector systems at various operating conditions determined from a non-linear steady-state system model simulated with EES (F-Chart Software, 2003). In particular, the influence of an IHX on the performance of a CO_2 ejector system is investigated and the resulting COPs and cooling capacities are compared to conventional transcritical CO_2 systems. Furthermore, the influence of the performance maximizing high-side pressure, an independent operational parameter in conventional transcritical CO_2 systems (Inokuty, 1928), is studied for the CO_2 ejector systems and high-side pressure control correlations are derived.

2. ANALYSIS

Four different mobile air-conditioning systems were modeled based on sets of approximately 2500 generally non-linear time-independent equations solved simultaneously with EES. Two of those were conventional transcritical CO_2 systems (MAC and MACI). The other two systems had an ejector rather than an expansion valve (MACE and MACEI). Both the MACI and MACEI model had an IHX. The layout of the most complex system among those four is shown in Figure 1 (MACEI). A corresponding Ph-diagram for a typical operation condition of this setup is shown in Figure 2.

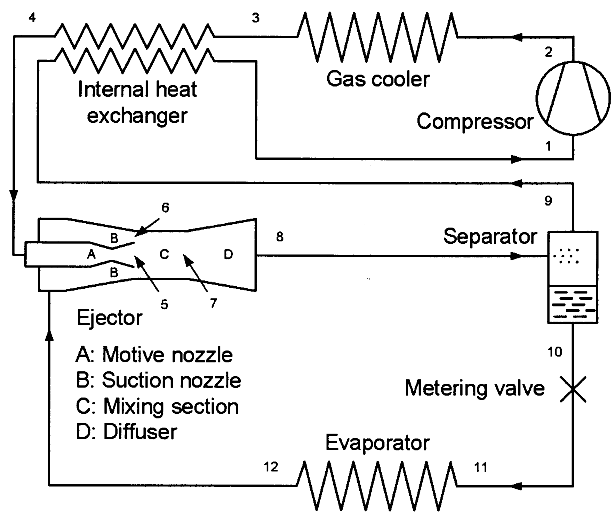

Figure 1: Transcritical CO_2 system with ejector and IHX (MACEI)

Table 1 summarizes component specifications used to simulate the four different systems. The microchannel tube heat exchanger capacities were determined from LMTD calculations. Compressor efficiencies were taken form curve fits obtained from actual experiments. The model validation for the MACI system as well as correlations used for heat transfer and pressure drop is described in more detail by Yin *et al.* (2001) and by Elbel and Hrnjak (2003). The iterative ejector calculations were implemented according to the analysis carried out earlier by Kornhauser (1990), based on conservation of energy and momentum. The pressure in the mixing section was assumed to be a constant independent variable in each simulation run.

Thermodynamic non-equilibrium effects as well as kinetic energies at the ejector inlets and outlet were not taken into account. The ejector was treated adiabatically. The isentropic efficiencies of its three major components were modeled as shown in equations (1) – (3).

Motive nozzle
$$\eta_{\text{motive nozzle}} = \frac{h_{\text{motive flow inlet}} - h_{\text{motive nozzle outlet}}}{h_{\text{motive flow inlet}} - h_{\text{motive nozzle outlet,isen}}} \quad (1)$$

Suction nozzle
$$\eta_{\text{suction nozzle}} = \frac{h_{\text{evaporator outlet}} - h_{\text{suction nozzle outlet}}}{h_{\text{evaporator outlet}} - h_{\text{suction nozzle outlet, isen}}} \quad (2)$$

Diffuser
$$\eta_{\text{diffuser}} = \frac{2 \cdot \left(h_{\text{diffuser outlet}} - h_{\text{mixing section, isen}}\right)}{u^2_{\text{mixing section}}} \quad (3)$$

Table 1: Major components used for the simulation of a mobile air-conditioning system

Compressor	Type	Reciprocating
	Displacement [cm^3]	Variable (33 @ max)
Expansion device		Needle valve / Ejector
Gas cooler	Description	Microchannel brazed Al tubes, 1 pass, 3 slabs, cross-counter flow
	Face area (width x height) [cm^2]	60.8 x 34.9 = 2122
	Core depth [cm]	2.03
	Core volume [cm^3]	4307
	Air side surface [m^2]	7.1
	Free flow cross-sectional area [m^2]	0.1617
	Refrigerant-side surface area [m^2]	0.53
Evaporator	Description	Microchannel brazed Al tubes, 24 pass, 2 slabs, cross-counter flow
	Face area (width x height) [cm^2]	24.4 x 17.6 = 430
	Core depth [cm]	8.5
	Core volume [cm^3]	3655
	Air side surface [m^2]	4.4
	Free flow cross-sectional area [m^2]	0.0315
	Refrigerant side surface area [m^2]	0.92
IHX	Description	Brazed microchannel tubes, counter-flow
	Length [m]	0.444

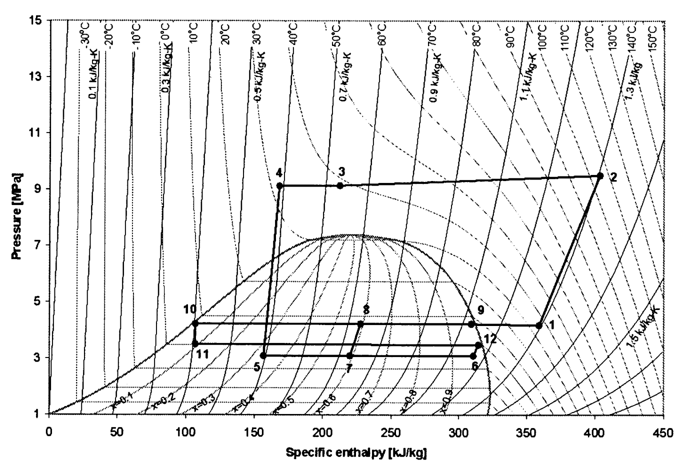

Figure 2: Ph-diagram for a transcritical CO_2 system with ejector and IHX (MACEI)

The evaporator cooling capacity and COP calculations based thereon were corrected for losses caused by the fan power terms according to equations (4) and (5).

Cooling capacity $$\dot{Q}_{system} = \dot{Q}_{evaporaor} - \dot{W}_{indoor\ fan} \quad (4)$$

COP $$COP_{system} = \frac{\dot{Q}_{evaporator} - \dot{W}_{indoor\ fan}}{\dot{W}_{compressor} + \dot{W}_{indoor\ fan} + \dot{W}_{outdoor\ fan}} \quad (5)$$

All simulations were carried out with an air inlet temperature to the evaporator of 35°C and 40% relative humidity. For numerical convergence reasons, the refrigerant qualities at the evaporator exit and the vapor port of the adiabatic liquid-vapor separator were assumed to be 99%, whereas the liquid port quality was taken to be 1%.

The air flow rates across the evaporator and the gas cooler were 425m^3/hr and 3058m^3/hr, respectively. The isentropic efficiencies of the ejector nozzles and diffuser were fixed at 90%. The compressor speed was 1500rpm in all runs unless otherwise stated. The mixing pressure, the compressor discharge pressure and the air inlet temperature to the gas cooler were the independent parameters varied throughout the course of this investigation.

3. RESULTS AND DISCUSSION

Kornhauser's (1990) comparison of two idealized systems, one of which being a conventional system and the other being a modified system comprising an ejector was based on a simple thermodynamic cycle analysis. For given operating conditions, the maximum possible COP improvement was found to be a strong function of the mixing pressure in the mixing section of the ejector. In order to verify these results, the analysis was implemented in EES and the COP improvement potentials found for R12 were compared to those of CO_2 at the same ambient conditions. However, for the supercritical gas cooling process it was assumed that the refrigerant temperature at the exit of the heat exchanger was equal to the ambient outdoor temperature, which is the equivalent to an ideally operating condenser.

At an ambient outdoor temperature of 30°C and an evaporation temperature of -15°C, the analysis showed that the ejector system yielded an up to 21% larger COP than the conventional system with R12 as the working fluid and isentropic ejector nozzles and diffuser (Kornhauser, 1990). For the same conditions, the COP of the CO_2 ejector cycle was found to be 53% larger than that of the standard CO_2 system, revealing the dramatic COP improvement potential of the ejector when used in high-pressure CO_2 systems. The theoretical improvement potential of the CO_2 ejector system exceeded even 70% for increased evaporation temperatures (5°C) and higher outdoor ambient temperatures (45°C) in combination with higher compressor discharge pressures (15MPa) due to the increased pressure energy contained in the motive flow. Even though these results indicate that the use of an ejector is more beneficial to CO_2 systems than it is to systems working with CFCs / HCFCs / HFCs, they are of course unrealistic to some extend. Besides the assumption of an isentropic ejector operation, many idealizations on which the thermodynamic cycle analysis was based on are responsible for letting the results appear too promising. Moreover, the influence of the performance maximizing high-side pressure is not properly reflected in a simple thermodynamic cycle analysis.

More realistic results are obtained when the non-linear steady-state system model is used to simulate both the conventional and the ejector systems. The COP as a function of the compressor discharge pressure of the four different systems is shown in Figure 3. The air inlet temperature at the gas cooler inlet was chosen to be 35°C. For each compressor discharge pressure investigated, the ejector systems were simulated with different ejector mixing pressures in order to identify the performance maxima.

As expected, the graphs show that the high-side pressure, being an independent parameter in a conventional transcritical CO_2 system, can also be used to maximize the performance of CO_2 ejector systems. Provided the systems are operated at their COP maximizing compressor discharge pressures and at a fixed compressor speed, the comparison between MACEI and MACI shows that the use of an ejector yields an approximately 10% larger COP for otherwise identical operating conditions. The comparison also reveals that the COP maximizing high-side pressure is generally lower for systems equipped with an IHX.

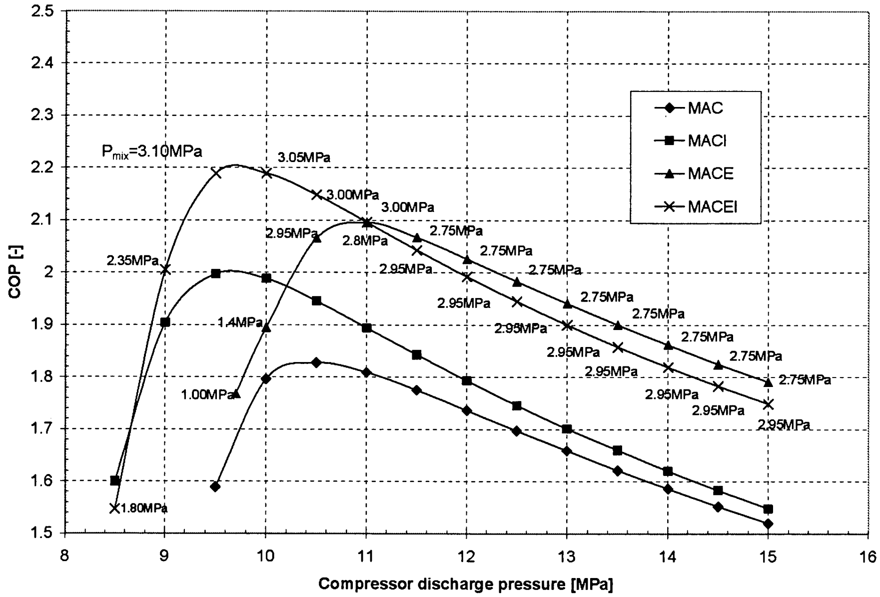

Figure 3: COP as a function of the compressor discharge pressure for 35°C gas cooler air inlet temperature

The cooling capacities of the four different systems for the same test condition are plotted in Figure 4. A comparison between MACEI and MACI shows that the use of an ejector increases the system capacity by approximately 15% provided the systems are operated at their capacity maximizing high-side pressures.

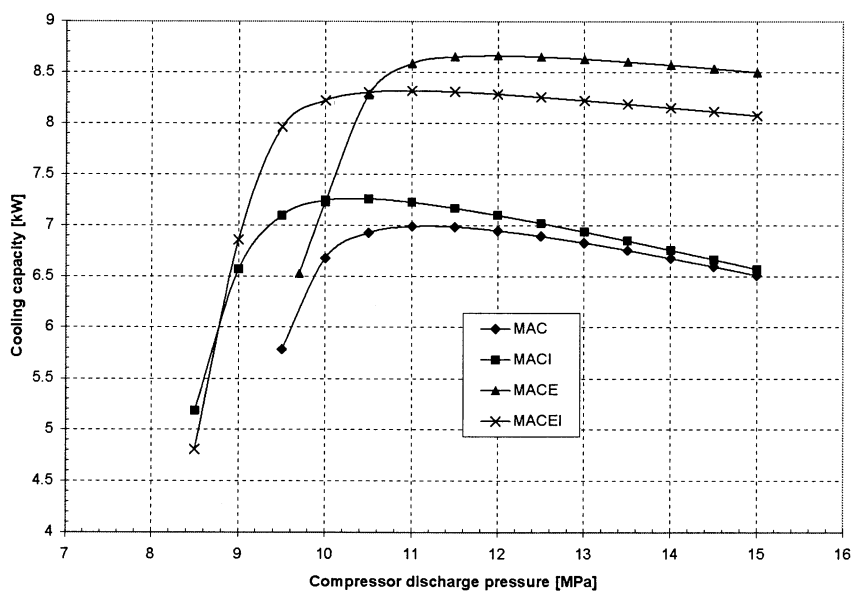

Figure 4: Cooling capacity as a function of the compressor discharge pressure for 35°C gas cooler air inlet temperature

However, a comparison between MACE and MACI reveals that the increase in cooling capacity can be as high as approximately 19% in the case of an ejector system without IHX. These results are somewhat counter-intuitive and require a more detailed explanation. In a conventional system, the use of an IHX always increases the cooling capacity, because the available phase change enthalpy difference across the evaporator is increased. This increase is achieved by an increased amount of compressor work due to higher superheats at the compressor inlet. Depending on the fluid characteristics, the additional amount of compressor work required can be over-proportionally smaller than the capacity gain and thus, for some fluids (e.g. CO_2, R134a), the use of an IHX increases the COP as well. However, the mechanism of an IHX in an ejector system works differently. Because of the vapor-liquid separator, the available phase change enthalpy difference across the evaporator a given saturation pressure is the same for an ejector system with or without IHX. This assumes that the metering valve downstream the liquid port of the separator is used to maintain a saturated vapor condition at the evaporator exit.

The ejector system without IHX has higher motive flow enthalpies resulting in larger velocities at the exit of the motive flow nozzle with an increased potential of entraining evaporator flow and re-compressing the mixed fluid streams to a higher diffuser exit pressure. In addition, the motive flow rate further increases because of the higher suction density resulting from a smaller superheat at the compressor suction port. Thus, among the four different systems, the highest cooling capacity for a certain test condition was always achieved with the MACE system. This system is distinguished from the other systems because of its tendency of having the lowest evaporation pressure, the highest diffuser exit pressure (highest pressure lift) and the highest motive flow rate. If an IHX is added to the ejector system, the motive flow enthalpy decreases leading to a lower refrigerant quality at the diffuser exit. Since the diffuser exit quality is equal to the ratio of the motive mass flow rate to the diffuser mass flow rate, the motive flow rate decreases in IHX ejector systems, provided the evaporator mass flow rate stays approximately constant. The results are: lower velocities in the ejector nozzles associated with lower diffuser exit pressures, and higher evaporation pressures (smaller pressure lift). This explains the decreased cooling capacity of the IHX ejector system. However, the decreased compressor mass flow rate reduces the required compressor work for a fixed speed of 1500rpm over-proportionally in comparison to the loss of cooling capacity. This eventually increases the system COP at a compressor discharge pressure being lower than the COP maximizing high-side pressure of the ejector system not having an IHX.

In the previously described simulations, both of the ejector systems resulted in higher cooling capacities and higher COPs at the same time when compared to the standard MACI system. It was already found that the maximum cooling capacity for this operating condition can be achieved with the MACE system (+19%), but the maximum possible COP increases still remain to be determined. In particular, it is not yet clear if the additional gain of cooling capacity of the MACE system can be traded for a COP improvement being higher than that of the MACEI system. This can be done by matching the given MACI cooling capacity (7.3kW) with the ejector system by allowing variable speeds for a fixed displacement compressor. The high-side pressures were adjusted according to the findings presented in Figure 3. For this case and a mixing pressure of 3.4MPa, MACEI had a 25% higher COP than the MACI system due to the reduced compressor speed of 1500rpm to 1190rpm. However, at matched cooling capacities, the COP of the MACE system, operating at a mixing pressure of 3.3MPa and a reduced compressor speed of 1020rpm, was 26% larger than that of MACI. This important finding shows that the use of an IHX is not favorable in a CO_2 ejector system equipped with either a variable speed or variable displacement compressor for situations in which a given cooling capacity of a standard system has to be matched. However, at a given compressor speed or displacement, the IHX can improve the COP of a CO_2 ejector system in comparison to its counterpart not having an IHX.

Another interesting finding is related to the choice of the mixing pressure in the ejector mixing section. While the simple cycle analysis predicts that the maximum system COP is obtained by adjusting the mixing pressure in such a way that the flow velocities at the exits of the ejector suction and motive nozzles are equal, this was not found to be the case in this analysis when the more realistic system model was used. Theoretically, with equal nozzle velocities, the shearing between the two streams to be mixed is at its minimum (Kornhauser, 1990). However, the steady-state system model shows that inefficiencies occurring in the mixing section can be compensated, and thus the maximum COP does not necessarily have to occur at mixing pressures yielding equal nozzle velocities as illustrated in Figure 5.

Generally speaking, the COP is not strongly affected by the mixing pressure. However, in accordance with Figure 3, the maximum COP occurs at a mixing pressure of 3.1MPa, at which the motive nozzle and suction nozzle velocities vary significantly from each other. The flow velocity in the mixing section has to be in between the two and is determined by the conservation of linear momentum.

Simulations were carried out for the four different systems at different outdoor temperatures (25°C, 35°C, 45°C) leaving all other parameters unchanged. From these results, high-side pressure control correlations for achieving maximum COP and cooling capacity, respectively, were derived for each of the four systems. In order to achieve generalized applicability of the correlations, the variable to be controlled was chosen to be the gas cooler exit pressure rather than the compressor discharge pressure. Thus, the pressure drop across the gas cooler was taken into account for the derivation of the correlations. For the same reasoning, the refrigerant temperature at the gas cooler exit was chosen in the linear correlations rather than the ambient outdoor temperature. Hence, the temperature glides (approach temperatures) of the imperfect heat exchange in the gas cooler were taken into account as well. The results are summarized in Table 2.

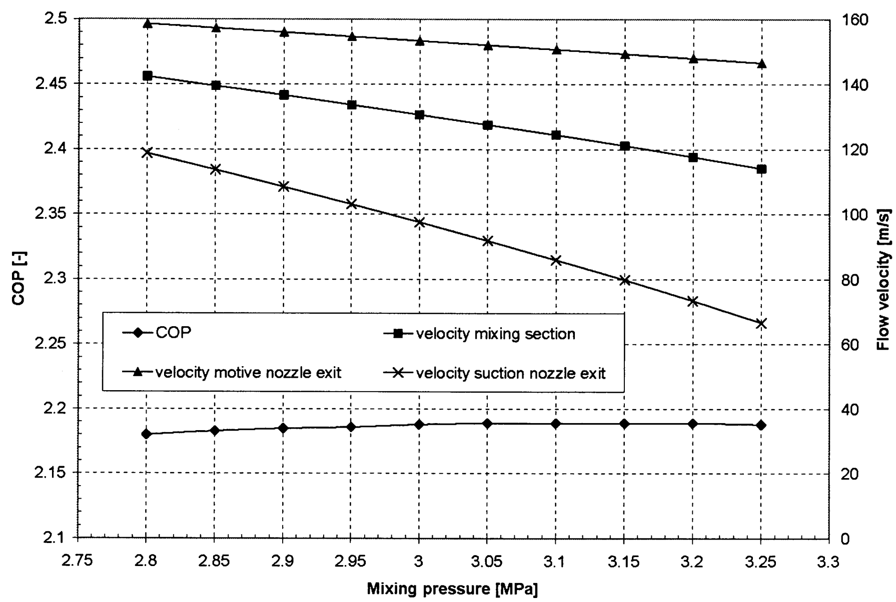

Figure 5: Influence of the mixing pressure on COP and ejector velocities at 9.5MPa compressor discharge pressure and 35°C gas cooler air inlet temperature (MACEI)

Table 2: Gas cooler exit pressure (P [MPa]) control correlations as linear functions of the refrigerant temperature (T [°C]) at the gas cooler exit

System	Correlation for max. COP	Correlation for max. cooling capacity
MAC	P = 0.2810 * T - 0.1959	P = 0.2451 * T + 1.9838
MACI	P = 0.2084 * T + 1.6029	P = 0.1907 * T + 3.2089
MACE	P = 0.2379 * T + 1.5274	P = 0.2858 * T + 1.2603
MACEI	P = 0.1609 * T + 3.6104	P = 0.2403 * T + 2.2020

The correlations found numerically still need to be verified, in particular those for the two ejector systems. Nevertheless, the correlations found are believed to be helpful approximations for the experimental determination of the performance maximizing high-side pressures.

4. CONCLUSIONS

The cooling capacities and COPs of transcritical CO_2 ejector systems were investigated using a steady-state comprehensive system model. The model was experimentally validated against a conventional transcritical mobile air conditioning system for a typical mid-sized car. The results are compared to those of conventional CO_2 systems without ejector. Due to limited availability of experimental ejector data, the focus of this study was on the trends of the results rather than on their exact numerical values.

In particular, it was found that the high-side pressure of a transcritical CO_2 ejector system, very much like in conventional system, represents an independent variable used to maximize either the cooling capacity or the COP. A set of general applicable high-side pressure control correlations were derived. These can be used to approximate the actual performance maximizing high-side pressures in future ejector system experiments.

The mechanism of an IHX in an ejector system differs greatly from that in a conventional system. It was shown that an IHX in an ejector system, unlike in a conventional system without ejector, does not necessarily increase the cooling capacity, i.e. in this comparison, the ejector system without IHX resulted in having the largest capacities for the test conditions considered. Furthermore, it was shown that in simulations with matching cooling capacities of a conventional system, the ejector system without IHX even showed higher COPs in comparison to the ejector system with IHX. However, for a given compressor speed, the ejector system with IHX resulted in the largest COPs, because of the trade-off between increased COP and increased capacity of the ejector system without IHX.

NOMENCLATURE

COP	: Coefficient of performance [-]
h	: specific enthalpy of refrigerant [kJ/kg]
IHX	: internal heat exchanger
MAC	: conventional mobile air-conditioning system
MACE	: mobile air-conditioning system with ejector
MACEI	: mobile air-conditioning system with ejector and IHX
MACI	: conventional mobile air-conditioning system with IHX
P	: refrigerant pressure [MPa]
$\dot{Q}$	: cooling capacity [kW]
T	: refrigerant temperature [°C]
u	: flow velocity [m/s]
$\dot{W}$	: power [kW]

<u>Greek</u>

η	: isentropic efficiency

<u>Subscript</u>

isen	: isentropic

REFERENCES

Denso Corporation Kariya, Japan, 2002, Ejector Cycle System, *U.S. Patent No. 6,438,993B2*

Engineering Equation Solver-Academic Version 6.881, 2003, F-Chart Software, Middleton, WI

Elbel, S.W., Hrnjak, P.S., 2003, Experimental and Analytical Validation of New Approaches to Improve Transcritical CO_2 Environmental Control Units, University of Illinois at Urbana-Champaign, *ACRC Report CR-52*

Gay, N.H., 1931, Refrigerating System, *U.S. Patent No. 1,836,318*

Harrell, G.S., Kornhauser, A.A., 1995, Performance Tests of a Two-Phase Ejector, *Proc. 30th IECEC*, Orlando, FL, p. 49-53

Inokuty, H., 1928, Graphical Method of Finding Compression Pressure of CO_2 Refrigerating Machine for Maximum Coefficient of Performance, *Proc. 5th Int. Congress of Refrigeration*, Rome (Italy), p. 185-192

Kornhauser, A.A., 1990, The use of an Ejector as a Refrigerant Expander, *Proc. 1990 USNCR/IIR-Purdue Refrigeration Conference*, West Lafayette, IN, p. 10-19

Yin, J.M., Bullard, C.W., Hrnjak, P.S., 2001, R-744 Gas Cooler Model Development and Validation, *Int. J. Refrig.*, vol. 24, p. 652-659